DESIGN, CHARACTERIZATION, AND CONTROL OF THE NASA THREE DEGREE OF FREEDOM REACTION COMPENSATION PLATFORM

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Introduction

Increasing research is being done into industrial uses for the microgravity environment aboard orbiting space vehicles. However, there is some concern over the effects of reaction forces produced by moving objects, especially motors, robotic actuators, and astronauts. Reaction forces produced by movement of these objects may manifest themselves as undesirable accelerations in the space vehicle, making the vehicle unusable for microgravity applications. It is desirable to provide compensation for such forces using active means.

This paper presents the design and experimental evaluation of the NASA three degree of freedom reaction compensation platform, a system designed to be a testbed for the feasibility of active attenuation of reaction forces caused by moving objects in a microgravity environment. Unique "linear motors", which convert electrical current directly into rectilinear force, are used in the platform design. The linear motors induce accelerations of the displacer inertias. These accelerations create reaction forces that may be controlled to counteract disturbance forces introduced to the platform. The stated project goal is to reduce reaction forces by 90%, or -20 dB. Description of the system hardware, characterization of the actuators and the composite system, and design of the software safety system and control software are included.

System Hardware

Figure 1 shows the design of the platform system. The platform system consists of a passive spring-mass-damper with added active components and sensors. The passive system attenuates forces at frequencies greater than the resonance, and passes forces at frequencies below the resonance. Figure 2 shows a Bode plot of the transfer function from the disturbance force applied to the platform to the residual force felt at the mechanical ground. Since the passive system provides at least -20 dB disturbance attenuation for frequencies above 88 rad/s, the active system design should be most concerned with disturbance rejection below that frequency. The resonant frequency could be lowered by decreasing the spring constant, at the expense of larger platform excursion, or by increasing the system mass, which may not be desirable in a space-going system. Also, damping could be added to reduce the effect of the resonance, but this may spread the phase transition over an unacceptably large frequency range.

The displacers of the linear motors are constrained to vertical motions with respect to the platform, and can thus react to vertical disturbance forces (along the

z-axis) and moments about the x- and y-axes. The motors are each capable of 712 N maximum force. All have a displacer mass of 5.6 kg, and a stroke of 0.3 m. Some insight can be gained by using the maximum force rating of the motors and the stroke limit to plot force and position attainable as a function of frequency, as shown in Figure 3. Below 4.8 Hz, the force available is limited by the position constraint; above that frequency, the position amplitude is limited by the maximum force constraint. Therefore, it is safe to attempt control at high frequencies, while commanding a large-amplitude control signal at low frequencies may be unsafe or ineffective. The switch frequency could be decreased by increasing the mass of the motor displacer, which may be undesirable, or by increasing the displacement limit, which would require replacing the motors. Increasing the motor mass would have the added effect of decreasing the maximum velocity, which would decrease forces due to friction and back-EMF.

All of the motors are equipped with optical incremental encoders accurate to 10 μm, home switches, and limit overrun switches. In addition, each motor is equipped with a compressed air "spring" support system to counteract forces due to gravity on the displacers. Maximum velocity of the motor displacers for sinusoidal force inputs is 4.2 m/s.

The force sensors and accelerometers are piezoelectric and are effectively high-pass filtered with a time constant of 2.5 s due to their design, making control of low frequencies using these sensors impossible. The force sensors have a maximum rating of 2670 N, and the accelerometers have a maximum rating of 98 m/s^2.

Communication between the control program and the motors and sensors takes place through a Programmable Multi-Axis Controller (PMAC) board. This board does encoder interpretation and velocity estimation for the motors, receives information from the sensors, performs commutation for the three-phase motors, and sends current commands generated to the motors. Motor force commands are sent out at 2.3 kHz. The board also performs auto-shutdown of the motors in case of a position limit fault. The PMAC board has a built in high-level motion control language, which is interpreted in real time rather than being compiled; this makes program execution very slow, and unsuitable for running extensive control programs.

The actual control takes place on a 80486-based PC running at 33 MHz. The control program is written in C, and compilation is optimized for speed by using some of the features of the 80486 microprocessor. The control loop runs at 1.1 kHz.

Characterization
Without accurate modeling of motor and composite system behavior, high-performance control is not possible. In particular, information on the force constant, mass, friction, maximum force and velocity, and bandwidth of each motor are needed before any active compensation using the motors can be attempted. Although the motors have electrical and mechanical characteristics very similar to

three-phase rotary motors, the mechanical stops prevent the use of rotary motor characterization techniques. Instead, techniques similar to those utilized in robotics were used to prevent motor damage[1]. These methods use small cyclical forces or motions to obtain data on motor parameters.

During the characterization, it became apparent that there were some dynamics in the motor and/or the air spring that had not been accounted for. Further examination revealed the presence of a position-dependent force offset. This offset requires that, at a certain position, the motors must exert a constant force to prevent the motor displacers from accelerating. The offset is probably the result of a "detent force," an attraction of the motor displacers to certain positions along their tracks, plus position-dependent air spring dynamics. The data taken for one of the motors, and the function used to model this phenomenon, are shown in Figure 4. The modeling function takes the form of a sinusoid-plus-slope-plus-constant.

Control
The control consists of three discrete parts: the force feed-forward controller, which directly responds to incoming forces read from the force sensors; the acceleration feedback controller, which responds to accelerations of the platform mass; and the motor position controller, which attracts the motors to equilibrium position, provides software damping for the motors, and also acts as a primary safety system.

The feedforward force control is a very straight-forward design, similar in principle to methods used in audio noise reduction. The disturbance forces are obtained by the force sensors; the signals are then negated (phase inverted) and reapplied using the actuators. Performance is limited by the design of the force sensors, motor modeling errors, and the digital delay inherent in all digital systems. Although only preliminary data has been collected on this control scheme, simulations have shown that 20 dB attenuation is achievable for frequencies between 55 $^{rad}/_s$ and 150 $^{rad}/_s$.

Control of the platform using feedback of the acceleration data proved to be a difficult problem. Phase shifts due to the platform itself, the piezoelectric nature of the sensors, and the time delay inherent in digital systems combined to cause problems with stability and control bandwidth. Classical control methods would produce the desired disturbance attenuation at high frequencies only at the expense of disturbance amplification at low frequencies, and state-space control seemed encouraging in simulation, but was too sensitive to partly measured or unmeasured values.

It is necessary to have a motor position controller to attract the motors toward zero position, so that disturbances caused by the motor triggering the safety system are kept to a minimum; it is also desirable to have velocity control to provide damping. The proportional-derivative (PD) control scheme is well documented and seems suitable for this task, but closer examination reveals limitations in this scheme. In order to insure that the limits are never overrun, a PD-controller would

have to have a resonant frequency of about 36 $^{rad}/_s$, significantly degrading the lower frequency response of the combined controller.

To alleviate this problem, higher-order functions of position and velocity are used to achieve a bumper-like effect. These types of functions tend to have small effect at high frequencies or small amplitude motions, but large effect at low frequencies or high amplitude motions. This has the effect of allowing high frequencies, but attenuating low frequencies where the motor cannot exert full force safely. Careful selection of the gain parameters allows only slight degradation in frequency response of the force and acceleration controllers, while providing another level of safety for the motors and attracting motor displacers toward equilibrium position.

Unfortunately, operation of the nonlinear "bumper" is directly opposed to operation of the acceleration controller. Any control effort from the bumper shows up at the platform as an acceleration; if the acceleration controller is working properly, it will then attempt to cancel this acceleration by applying an opposing force, defeating the purpose of the bumper controller. This problem can be solved by including a reference term before the acceleration controller, that is a result of the bumper control effort filtered through the plant model to give an acceleration. See Figure 5.

In addition, superimposing the desired forces from all the controllers may result in a condition where the desired bumper force is defeated, leading to a motor collision and possible damage. To avoid this, the desired forces from the force sensor and accelerometer loops are filtered through a nonlinear function that is dependent on the desired bumper force. The forces are superimposed only if the sign of the combined force is the same as that of the bumper force; if the signs are opposite, the combined force is multiplied by a gain of between zero and one, depending on the magnitude of the bumper force. Lower gain is applied for higher bumper force, so that the bumper force takes higher precedence. This policy is summed up in the following equation: $F_{out} = F_b + f(F_b)F_c$, where F_b is the desired bumper force, F_c is the desired control force, and $f(F_b)$ is a continuous function which equals 0 for F_b greater than an upper threshold value, 1 for F_b less than a lower threshold value, and decreases linearly from 1 to 0 for values of F_b between the two threshold values.

Conclusions

The force and stroke limits of the motors both serve as actuator saturation limits. The force limit sets the saturation at high frequencies, while the stroke limit sets the saturation at low frequencies.

Classical control proved to be ineffective for control in the acceleration feedback loop. Control using classical methods yielded either small attenuation of forces or attenuation at high frequencies only at the expense of amplification at low frequencies. Also, the use of state-space methods in the acceleration controller proved to be ineffective due to oversensitivity to partly measured or unmeasured

quantities, and the inability of state-space controllers to accept reference inputs in the case of the platform system [2].

The nonlinear "bumper" position and velocity controller proved to be more desirable than the commonly-used PD controller due to the bumper's lower force commands for high frequency/low amplitude motor motion. This allowed greater bandwidth of the combined controller.

The anticipated force disturbance rejection for the combined system is at least -20 dB attenuation for frequencies greater than 55 $^{rad}/_s$, which will extend the lower bandwidth by 33 $^{rad}/_s$ below that of the passive system alone, without an increase in platform mass or decrease in spring stiffness.

References

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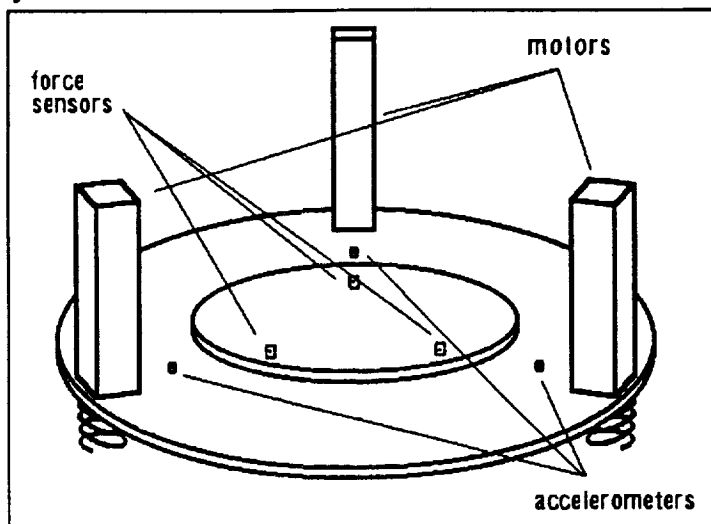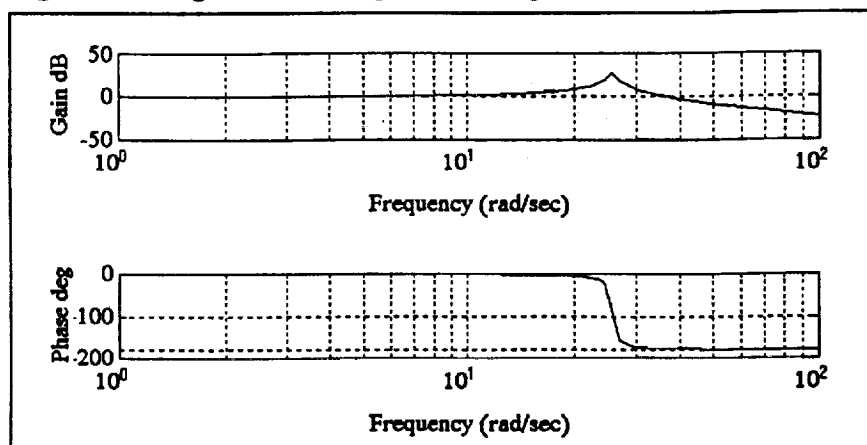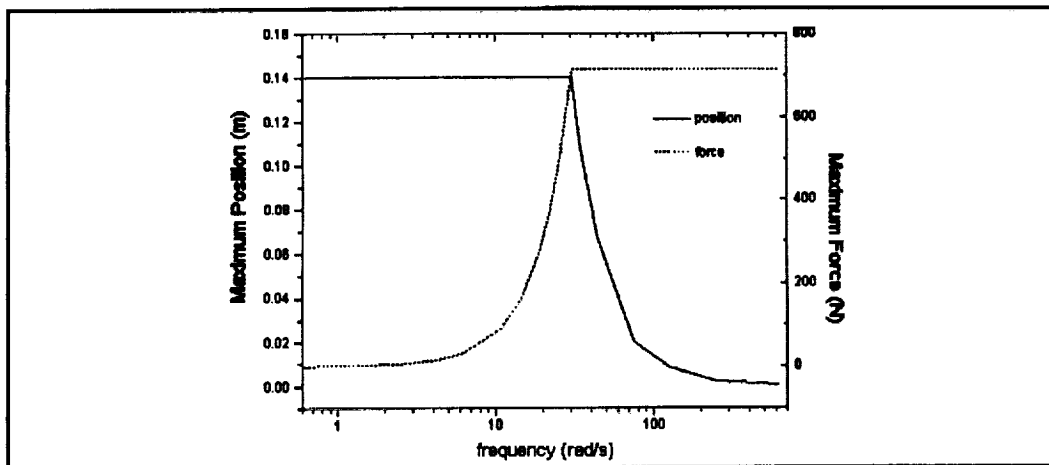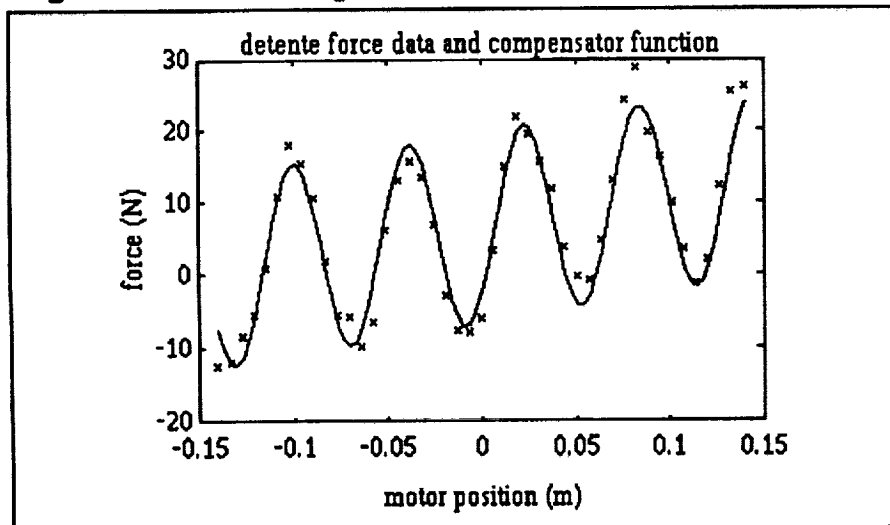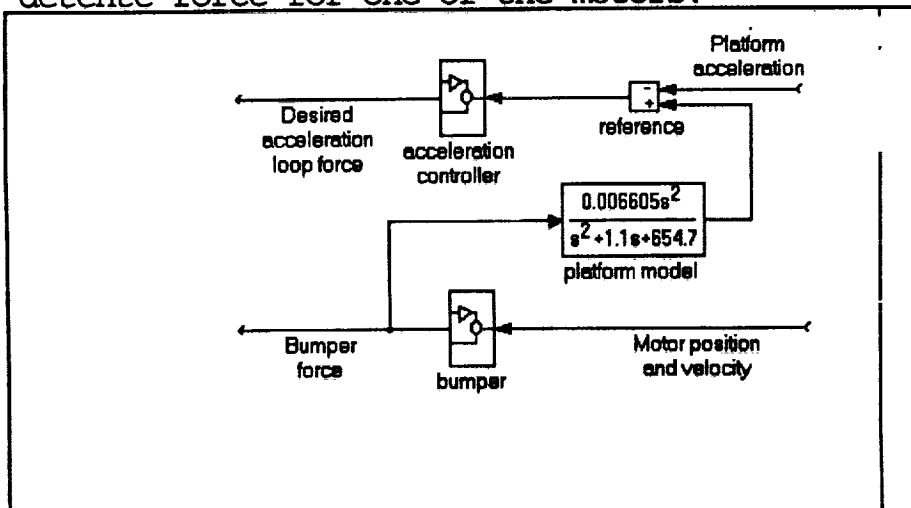
Figure 1 Diagram of the platform system.

Figure 2 Bode plot of the disturbance force to ground transfer function for the passive platform system.

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Figure 3 Maximum position and force versus frequency.

Figure 4 Characterization and model of the detente force for one of the motors.

Figure 5 Block diagram showing the correction for opposing acceleration and position control.